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Lubrication

A Technical Publication Devoted to the Selection and Use of Lubricants

THIS ISSUE

ENGINE OIL PRESSURE

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LUBRICATION

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ENGINE OIL PRESSURE

THE LIFE LINE of the human body is the blood system with the heart pumping the fluid of life through arteries, capillaries and veins to vital organs and parts of the body where it can best serve its purpose. Similarly the lubrication system with its oil pump, galleries, by-pass lines and orifices is the life line of an engine.

An important condition of the blood system that should be checked regularly is blood pressure since it is an indicator of how the system is working. Limits or ranges of blood pressure have been established that are considered normal, however, conditions may arise, such as extra body activity, that will cause variation in pressure beyond the normal blood pressure range. These variations are also normal and not detrimental but illustrate the point that consideration should be given to all the factors involved before conclusions are drawn regarding the significance of blood pressure changes. Similarly, in the lubrication system of an engine the oil pressure may be very useful for determining its proper functioning which is essential for long engine life.

The term "engine oil pressure" is often found in literature on engines, however, little is ever mentioned beyond stating a pound per square inch value at some speed condition for the particular engine involved. The question naturally arises as to why measure oil pressure continuously. The answer obviously is to note if oil pressure changes occur during engine operation; however, the significance of these changes can sometimes be misinterpreted. Therefore the purpose of this article is to accumulate information on engine oil pressure to better understand why it varies and the significance of this variation; also, to review some details that have a bearing on lubrication system design.

Engine Lubrication Systems

The lubrication system of an engine consists primarily of an oil pump and oil lines or galleries to feed bearings, timing gears, valve trains, cylinder walls and other lubrication points. To complete the system an appropriate relief valve is incorporated to confine the oil to the main galleries and to relieve peak cold oil pressures. If desired an oil filter may also be included in the system. For example, one of the most common lubrication systems, a complete pressure system, is shown as Figure 1.

A variation is found in a system that incorporates a separate oil reservoir, otherwise known as a dry sump engine, as shown in Figure 2. Two separate oil pumps or a double pump in one body are used in this type of system.

Some engine manufacturers successfully use a pressure system to all points except the connecting rod bearings and wrist pins. Connecting rod bearings are supplied with oil by equipping the rod bearing caps with dippers to scoop the oil from special troughs in the oil pan aided by impinging a stream of oil into the scoops by means of accurately aligned jets. In this system also it is obvious that oil flow rates are of major importance.

Large Diesels, as shown in Figures 3 and 3A, usually have two separate oil systems. One system independently lubricates piston and cylinder liner walls while a crankcase system is used for lubrica-

tion of the remainder of the engine. Two different oils are usually used in this engine. Also, a secondary system is used on these larger engines to carry the crankcase oil to independent filtration equipment.

Function of the Oil

In all engines the oil is called upon to perform more than one service. The primary service is to maintain a continuous oil film between moving surfaces to prevent metal-to-metal contact. In addition the oil is required to carry away heat generated by friction (oil shearing), act as a piston-to-cylinder wall seal and in some cases to increase piston cooling. It is quite obvious that the rate of oil flow is an important factor if the oil is to best serve its intended purposes.

Instrumentation

It is very desirable to incorporate in an engine lubrication system some method of measuring oil flow. However, instrumentation of this kind is costly and in most cases impractical, particularly in the automotive field. A variable closely related to fluid flow is pressure. Since pressure gauges are relatively less costly to make and easy to incorporate in an engine lubricating system they are used as a means of assuring the operator that engine lubricating oil is flowing. However, it is difficult to interpret pressure gauge indications as rates of oil flow since many factors affect oil pressure in an engine as will be pointed out. The need for a low cost instrument for measuring oil flow will become more apparent in the subsequent discussion.

Factors Affecting Oil Pressure

The factors affecting oil pressure should be considered before drawing conclusions regarding fluctuations in engine oil pressure. To better understand the significance of oil pressure gauge indications, consideration should be given to some fundamentals of fluid flow. Assume that the lubrication system consists of a fixed size and length of pipe (galleries) with a definite size opening for leakage (bearing clearances, squirt holes and by-pass) with fluid at a constant temperature supplied

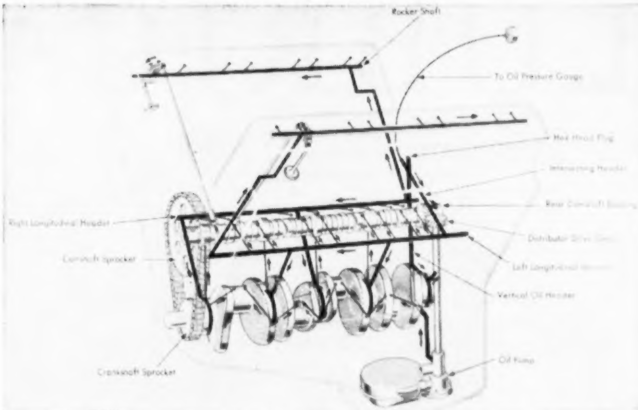
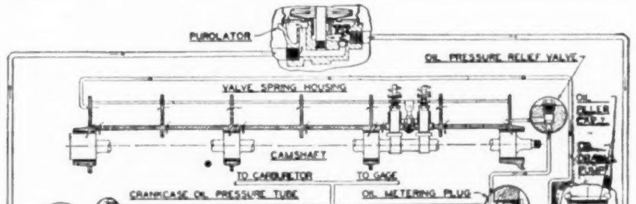


Figure 1 *Courtesy of Cadillac Motor Car Division, General Motors Corp.*



Courtesy of Scripps Motor Company

Figure 2

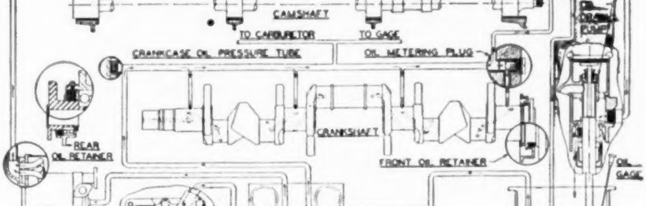
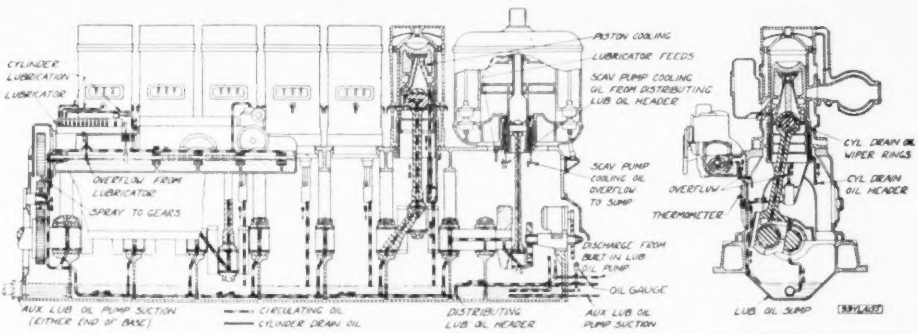


Figure 3 *Courtesy of Fairbanks, Morse & Co.*

Figure 1 — Details of the Cadillac engine lubrication system. Note this involves complete pressure by a single oil pump.

Figure 2 — Lubrication system of Scripps Motors. Note that in this dry sump type of engine separate oil pumps are used.

Figure 3 — Lubricating oil piping arrangement for a wet sump Diesel engine.



Courtesy of Fairbanks, Morse & Co.

Figure 3A — Lubricating oil diagram for a Fairbanks, Morse Diesel engine.

by a constant volume (displacement) pump, which is the most common type used in engine practice. Under these conditions a definite pressure gradient and discharge pressure at the system outlet is established in the system. The only characteristic of the fluid that will change this pressure condition is viscosity. The other factors that would change these pressures are of a mechanical nature (change in leakage by a change in discharge outlet size or volume supply) or in temperature which again would only affect viscosity. It is recognized that pump discharge volume must be controlled in such a system since changes in viscosity or pressures affect pump output volume.

Before studying the effects of these variables in a simplified oil system or in actual engines, consideration should be given to the equations of fluid flow. A simplified version of the flow equation for fluids in closed paths (pipes) under laminar flow conditions* is:

$$V = \frac{PD^2}{CN} \qquad (1)$$

where V = Flow (volume)
P = Pressure Change Along Flow Path
D = Pipe Diameter
C = Constant for Proper Dimensions
N = Viscosity

This equation states that laminar fluid flow in closed paths varies directly with pressure gradient and pipe diameter and inversely with viscosity. Also, a change in path diameter has a greater effect on the flow than either pressure drop or viscosity changes since flow varies as the square of the diameter. This equation is applicable in determining coefficients of discharge for some types of orifices and by-pass outlets.

Considering the positive displacement pump usually found in an engine lubrication system, its performance can be described by the following equation:

$$V = V_p - V_s - V_c \qquad (2)$$

Where V = Actual Delivery
Vp = Physical Pump Displacement Delivery
Vs = Slip Losses
Vc = Cavitation Losses

This equation merely states that the actual delivery is equal to the delivery based on the physical capacity and speed of the pump less slip and cavitation losses. The slip losses may be quite large since they are similar to laminar fluid flow between flat parallel plates. The equation for this condition is:

$$V_s = \frac{PBT^3}{12CNL} \qquad (3)$$

where Vs = Flow Between Close Flat Parallel Plates
P = Pressure Change Along Path
B = Width of Passage
T = Clearance Between Plates
C = Constant for Proper Dimensions
N = Viscosity
L = Length of Passage

This equation states that laminar flow between flat parallel plates varies with pressure gradient and viscosity in a manner similar to laminar flow in pipes, however, changes in clearance between plates, cubed functions, have a greater effect on flow than changes in pipe diameter. This equation may be applied in determining oil flow through bearing clearances.

The cavitation losses in an engine oil system under normal engine operating conditions are usually negligible. These losses can become detrimental, however, in cases of severe foaming or improper oil viscosity.

*Under normal conditions of operation, oil flow in an engine is of the laminar or viscous type and should not reach conditions of instability.

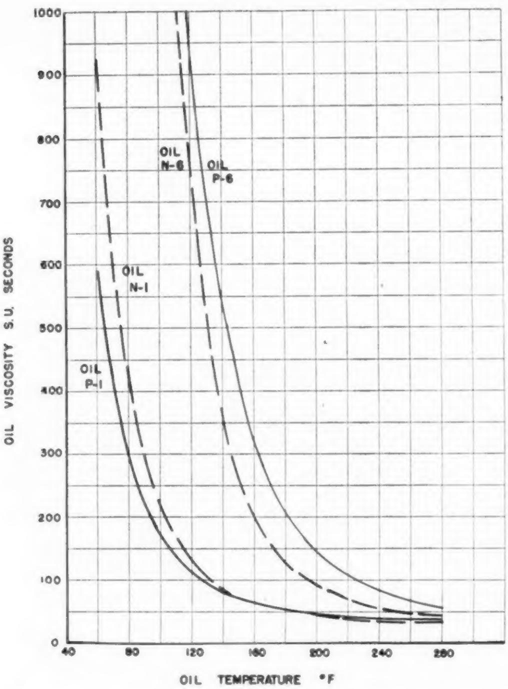


Figure 4 — Curves showing viscosity-temperature relationship. See text below for discussion.

Since oil viscosity varies inversely as a function of temperature and this fact will be considered in the subsequent discussion, the viscosity-temperature relationship of two pairs of oils, of widely different viscosity and viscosity index (see Table I), has been included as Figure 4 for information purposes. It may be noted that the relationship is not linear, which fact is important. In the lower temperature range large changes in viscosity are experienced with small changes in temperature while at the higher temperatures the reverse is true.

Laboratory Simulated Lubrication System

To study some variables associated with engine lubricating oil systems in simplified laboratory equipment the units shown in different views as Figures 5 and 5A were used. This equipment consists of two completely independent units. The schematic diagram of the individual units is shown as Figure 6.

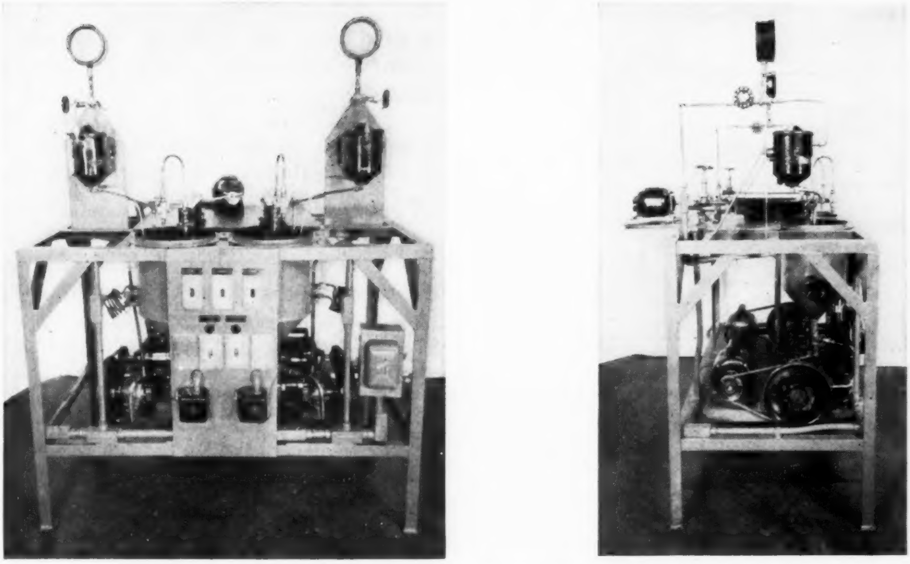
Each test unit consists, essentially, of a cylindrical reservoir with a conical bottom containing thermostatically controlled heating coils, a constant volume (gear) pump, a by-pass line with a throttling valve to control pressures in the system, a filter line and an accurately-metered contaminant extruder so that the unit can be used for evaluating oil filters. The extruder was not used for the work presented herein. The units are such that they can be readily modified. One such modification is shown as Figure 7. The screen filter shown in this setup was installed to insure removal of any extraneous contaminants that might lodge in valve passages, thereby varying the original restriction setting.

The oils used in this bench test type evaluator and in the engine tests to be discussed later are shown as Table I along with their viscosities and viscosity indices. It is pointed out that both naphthene and paraffin oils covering a wide range of viscosity grades and viscosity indices were included.

By using the test apparatus shown in Figure 7 with the throttling valve used to restrict flow we have essentially the pipe arrangement assumed earlier in this article. Tests were carried out using two oils P-3 and N-3S (Table I) whose Saybolt viscosities were approximately equivalent at 130°F. In the first test, pump speed was kept constant and oil temperatures at pump discharge were varied from 80°F. to 280°F. Essentially, oil viscosity was varied. The resulting pump discharge pressures and pressure drops through the line are shown as Figure 8. These curves show that discharge pressures and line pressure drops decreased with increased temperature (decreased viscosity). The low VI Oil N-3S showed substantially greater discharge pressures and line pressure drops than the high VI oil P-3 in the 80°F. region, but both oils were almost equivalent from 105 to 280°F. These results appear logical in view of the viscosity of the low VI oil being 600 SU seconds greater at 80°F. than that of the high VI oil while at 280°F. the low VI oil is only 5 SU seconds lower than the high VI oil. This may be seen from the temperature-viscosity relationship of these oils, shown as Figure 9. It

TABLE I
OIL VISCOSITY DATA

Oil Designation	*Viscosity, SUS At: 100°F.*	*130°F.*	*210°F.*	*Viscosity Index*
P-6	1680	712	121	87
P-5	1220	475	93	88
P-4	830	348	78	91
P-3	547	242	67	97
P-2	331	156	54	96
P-1	172	93	44	98
N-6	1690	510	78	<0
N-5	1178	384	69	<0
N-4	764	280	60	<0
N-3	565	218	55	8
N-2	315	135	48	19
N-1	205	100	44	20
P-3C	523	235	66	102
N-4A	704	280	66	63
N-3S	691	247	57	<0



Figures 5 and 5A — Laboratory equipment for studying engine oil pressure, see page 24 for discussion.

may be noted that the absolute* viscosities of the oils were equal at the respective oil temperatures where discharge and line drop pressures were equal.

The oil flow rates determined in this test are shown as Figure 10. The dimensionless quantity known as Reynolds Number (R), a parameter which characterizes the relative importance of viscous action, was computed for the severest condition based on viscosity of the oil at the pump discharge oil temperature. It was found on this basis that oil flow was laminar up to the points R indicated on the curves of Figures 8 and 10. These are the Reynolds Number values of 2000, otherwise known as the lower limit below which flow disturbances of any magnitude are eventually damped by viscous action. All flows were below the upper critical Reynolds Number value of 12,000 or, in other words, instability of flow was not encountered.

From the equation of flow (1) it can be seen that for constant flow and pipe size, line pressure changes and consequently discharge pressures vary with viscosity. Since viscosity varies inversely as a function of temperature, as shown in Figure 4, then, in this equipment the oil pressure-temperature relationship curves should follow the same trend as the viscosity-temperature curves. By comparing Figures 4 and 8 it may be noted that this was confirmed through the region of approximately constant flow.

Using the same unit in the second test, oil temperature was maintained constant at 130°F. and pump speed varied using oils P-3 and N-3S again. The pressure data obtained are shown as Figure 11. These curves show that discharge and line drop pressures increased with increased oil flow. Although the differences in respective pressure values between the two oils are considered to be of little significance, the relationship again showed lower discharge and line drop pressure for the lower viscosity oil at equal flow rates. These results agreed with the equation of flow. Since for any one discharge pressure the oil flow rates were not equal, even though the differences in flow rates are of no practical value, it is apparent that the oil pressure gauge reading alone did not indicate the quantity of oil flowing.

From Figure 12, where pump speed was varied with constant oil temperature and the viscosities of both oils were sufficiently high it is apparent that the system did not recognize oil VI and output was a linear function of speed. However, oil pressure increased with output as discussed above.

From these data it appears that high and low VI oils show little or no pressure difference at temperatures associated with normal engine operation.

The Reynolds Number was calculated for the maximum flow condition to determine what type of flow was being obtained. A value of 0.04 was obtained which, being less than 0.141, the critical value, indicated that laminar flow was experienced in all cases.

Actual Engine Oil Pressures

These simple tests indicate that normal equations of fluid flow are applicable to an engine lubrication

*Absolute viscosity is the product of the kinematic viscosity and specific gravity at the particular temperature involved.

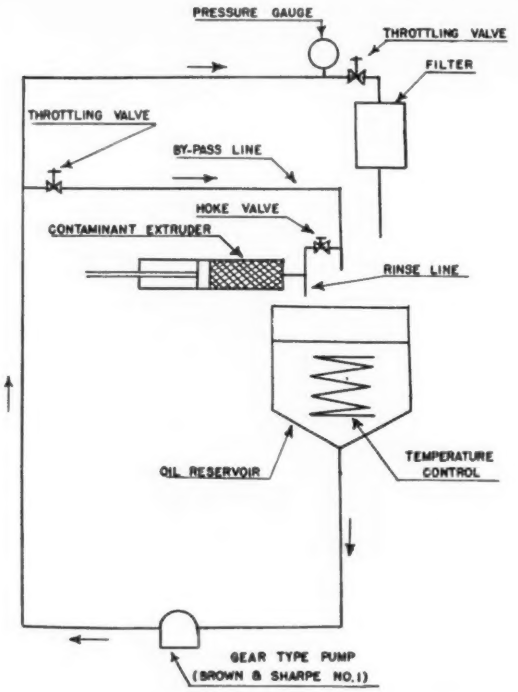


Figure 6 — Schematic diagram of an engine lubrication bench test. See page 24 for discussion.

system. Therefore, in an actual engine, oil pressures should be affected by the following factors:

1. Oil pressure should vary with oil viscosity or, since viscosity varies inversely as a function of temperature, oil pressure also should vary inversely as a function of oil temperature.

2. Oil pressure should vary with flow, or since pump output varies with speed, oil pressure should also vary with engine speed.

3. Oil pressure should vary inversely as a function of outlet size (oil passage clearances) in the system or, oil pressure also should vary inversely as a function of wear.

Some oil pressure data measured in an automotive type gasoline engine wherein all conditions were kept constant except for the indicated variable are shown as Figures 13, 14 and 15. It may be noted that these data confirm the oil pressure statements made above although the relationship trends are not exactly similar to those obtained in the simplified laboratory equipment.

Oil pressures as affected by engine speed in a two cycle high speed Diesel engine are shown as Figure 16. The engine in these cases was in perfect mechanical condition, operating conditions in each case were similar and two different model oil pumps were used. The same trend was shown by each pump, however, the newer model pump showed higher oil pressures which was expected since it was a larger capacity pump. In service when engines are overhauled caution should be exercised in obtaining the manufacturers' latest designed parts for the particular engine if the manufacturers' performance specifications are to be met.

Oil pressure-engine speed relationships were determined in a gasoline engine for one oil at various crankcase temperatures and a constant jacket temperature. The data obtained are shown as Figure 17. It is apparent that in this case oil viscosity based on crankcase temperature had little or no effect on oil pressure except at idling speeds although the engine speed effect was noted. At first, it would appear that perhaps the data are wrong or the equations of fluid flow do not apply. However, further data were obtained on this engine to better visualize what occurred in the oil system. Oil gallery temperatures were determined throughout the engine speed range at three different jacket temperatures with the crankcase oil temperature maintained as low as was possible (110-120°F.). The oil gallery temperatures are shown as Figure 18. It is quite apparent that jacket temperatures had a marked effect on controlling oil temperature in this engine. Other engines also have their particular characteristics relating jacket and oil temperatures. These data show that the oil temperature is not constant throughout the lubrication system, therefore, oil viscosity also varies throughout the system. The equations of flow are valid in the engine but become more complex to apply since oil viscosity varies along the flow path.

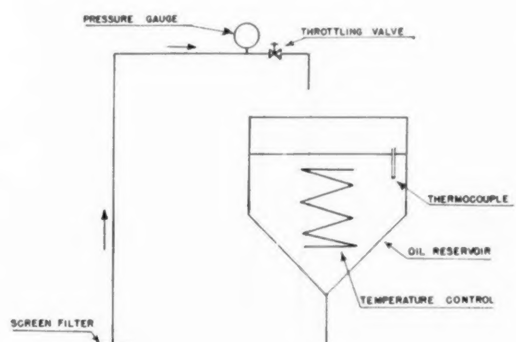


Figure 7 — Arrangement of a modified engine lubrication test system. See page 24 for discussion.

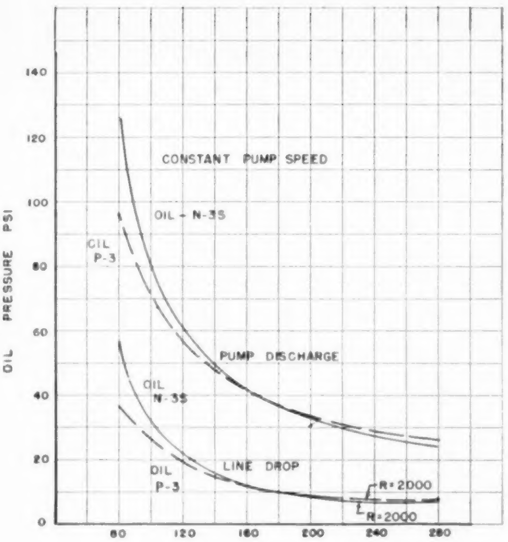


Figure 8 — Oil pressure-temperature relationship for a simulated lubrication system. See page 24 for discussion.

When oil pressure data such as the following obtained in a Diesel engine are encountered it would be unwise to adhere to the existing belief that low oil pressure always indicates defective lubrication.

Water Temperature, °F.	*Constant Speed Oil Pressure, PSI*
150	50
170	40
190	30

Operators who believe this and use a higher viscosity grade oil in an effort to increase oil pressure may not overcome their problem as will be shown subsequently.

The following data were obtained at constant speed and load conditions in a railway Diesel engine wherein the jacket coolant is used as the coolant for oil heat exchangers.

		Oil Pressure, PSI	
Jacket Temperature, °F.	*Oil Temperature, °F.*	*To Engine*	*Drop in Engine*
104	120	61	5
134	156	60	8
141	164	60	13
164	186	57	15
191	214	53	19
200	240	50	23

The effect of jacket temperature on oil temperature is apparent. Since temperature inversely affects viscosity the engine oil pressure reflected the reduction in viscosity. Although oil flow rates were not determined in this engine the increased pressure

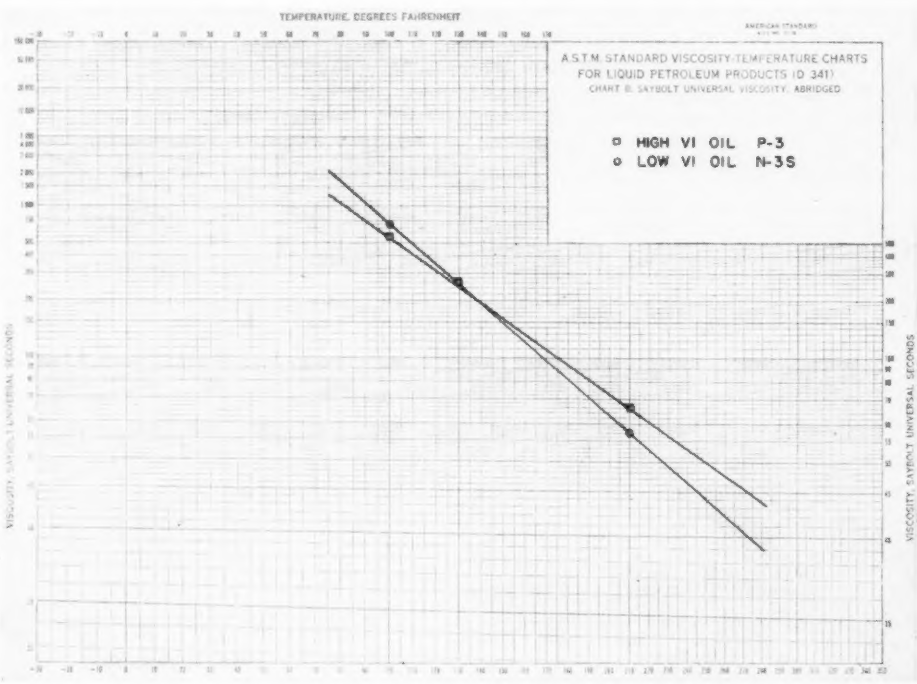


Figure 9 — Viscosity-temperature relationship for two typical oils. See page 24 for discussion.

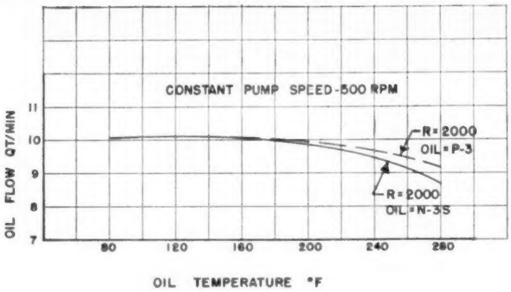


Figure 10 — Oil flow rates for a simulated lubrication system. See page 25 for discussion.

drops through the engine with decreased viscosity were associated with increased oil flow. These data show that the effect of jacket temperature on oil pressure is markedly different for different types of engines.

Oil pressures for all of the oils shown in Table I were determined in an automotive gasoline engine that was in good mechanical condition. Data were obtained at various speeds while other engine operating conditions were maintained constant. Then the oil pressure determinations were repeated using the same oils and the engine equipped with an oil pump that had .010" wear on the face of the drive gear and .008" increase in backlash between the gear teeth. The results obtained on the basis of oil pressure versus oil viscosity at 210°F. are shown as Figure 19.

It is apparent that with the normal pump at the various speeds shown, oil viscosity grade had very little effect on engine oil pressure. The greatest effect was shown at the highest speed and that was less than 4 psi over the whole viscosity range. The increase in pressure variation with increase in engine speed was associated with pump characteristics. Also, pressure was a function of viscosity rather than type of base stock or VI.

With the worn pump in the highest speed range (lower of upper pair of curves) a consistent reduction in oil pressure was experienced through all the SAE grades. This is reasonable since the excess ca-

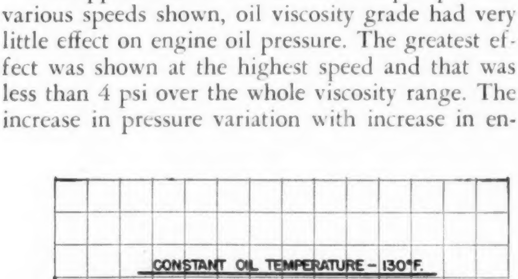


Figure 11 — Oil pressure-flow relationship for a simulated lubrication system. See page 25 for discussion.

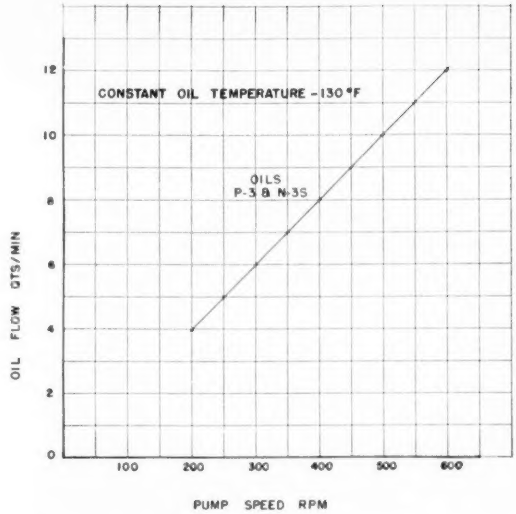


Figure 12 — Oil flow-pump speed relationship for a simulated lubrication system. See page 25 for discussion.

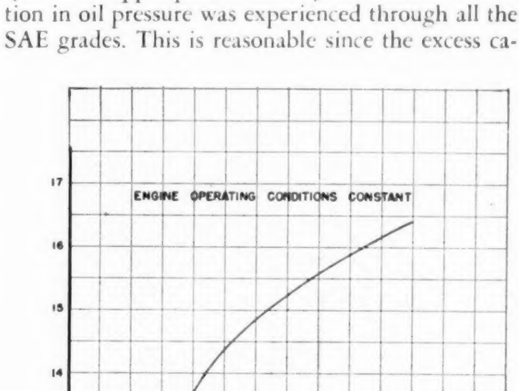


Figure 13 — Oil pressure-viscosity relationship for a gasoline engine. See page 26 for discussion.

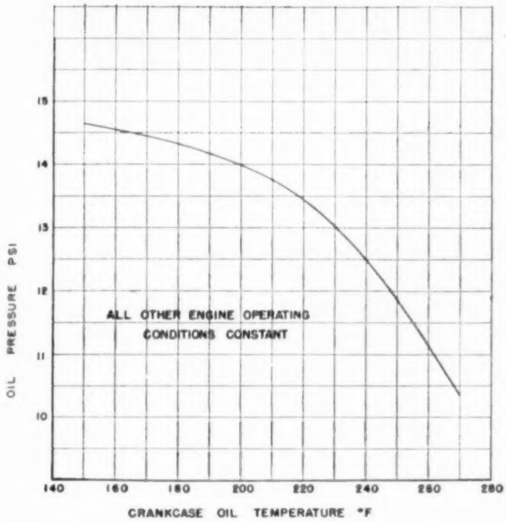


Figure 14 — Oil pressure-temperature relationship for a gasoline engine. See page 26 for discussion.

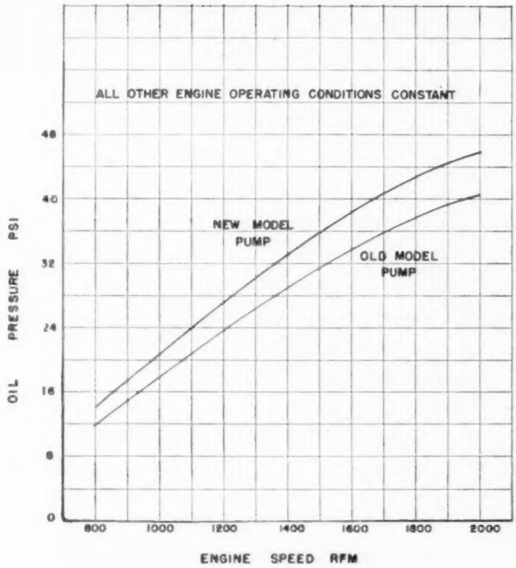


Figure 16 — Oil pressure-engine speed relationship for a Diesel engine. See page 26 for discussion.

pacity of the oil pump maintained the same relative order of pressures which reflected the slip loss in the pump resulting from the wider clearances. As speed was reduced, using the worn pump, oil pressures were lower and showed a further decrease with decreasing SAE grade. In the lowest speed range shown the possibility of insufficient flow rate for proper lubrication and cooling existed with oils below an SAE 40 grade since the manufacturer specifies a minimum of 25 psi oil pressure at these speeds.

These results indicate that at normal engine operating temperatures the effect of viscosity on oil pressure is relatively small when the engine and pump are in good mechanical condition. However, if large engine oil pressure changes are experienced with minor viscosity changes, it is an indication that an unsatisfactory mechanical condition exists.

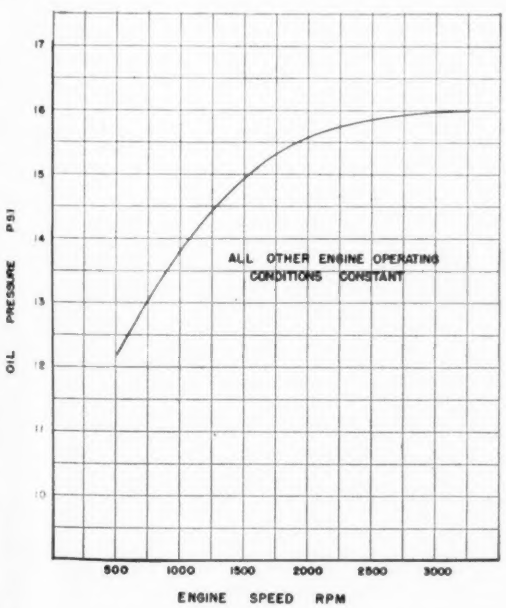


Figure 15 — Oil pressure-engine speed relationship for a gasoline engine. See page 26 for discussion.

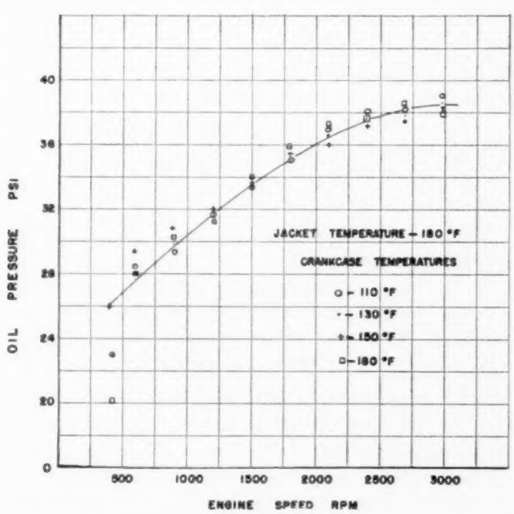


Figure 17 — Oil pressure-engine speed relationship for a gasoline line engine. See page 26 for discussion.

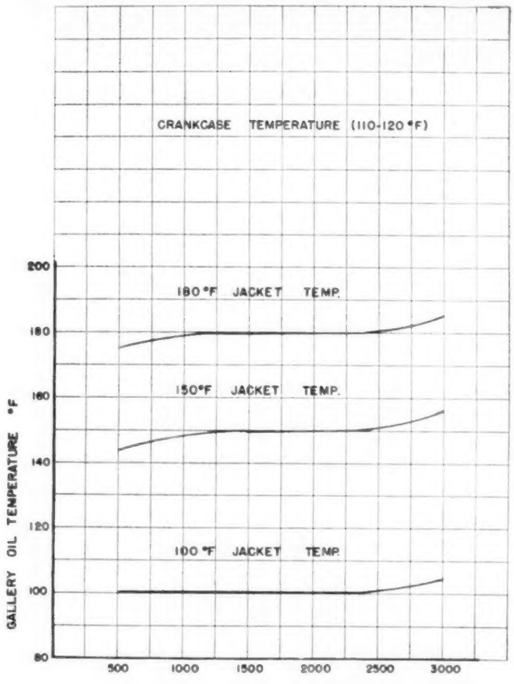


Figure 18 — Effect of jacket temperature on oil temperature in a gasoline engine. See page 26 for discussion.

Oil Pressure Related To Flow

That a reduction in oil pressure means an increase in flow in the engine system is usually accepted as fact, however, this is true only under certain conditions. Considering the oil pump, a reduction in outlet pressure increases volumetric efficiency, however, in the range of engine oil pressures normally experienced, this increase of efficiency is small and would increase flow only slightly. Considering the engine, the only factor that affects pressure is viscosity. However, in the normal operating range of crankcase temperatures there are only small changes in viscosity, therefore, also in pressure.

In the case where pressure drop is due to enlarged clearances, such as bearing wear, flow will increase through the enlarged clearances (Equation 3, page 23) but the total flow in the system remains the same except for the following condition.

Engine oil pumps are normally designed to have excess capacity to take care of normal engine wear or enlarged clearances. To avoid excessive pressures while the engine is new a by-pass arrangement controlled by oil pressure is incorporated in the system. If either a change in clearance or viscosity occurs that reduces pressure below the predetermined setting of the by-pass control, the by-pass valve closes diverting more oil to the engine galleries, assuming, of course, that the oil pump has excess capacity. The increased flow tends to restore oil pressure (Equation 1). If conditions arise that increase viscosity, or clearances are reduced, the process will reverse itself. Also, the condition can exist whereby the by-pass line is not sufficiently large to take care of all the excess flow; then the oil pressure gauge will indicate a pressure rise above the control setting which registers the restraint to increased flow (Equation 1, page 23).

In a worn engine or with a worn pump such as shown in Figure 19, it is apparent that at high speeds the pump still had sufficient excess capacity to override the control pressure setting of 30 psi, however, at lower speeds there was insufficient capacity to maintain pressure. The drop in pressure experienced did not mean more oil was flowing. Also, the reduced pressures may not have been a reflection of lower oil viscosity alone but also a reduction in flow (see Figure 10). In all of these cases it is apparent that the oil pressure gauge did not necessarily indicate the rate of oil flow.

Factors Affecting Viscosity

Oil viscosity in an engine is affected by fuel dilution and heat which reduce viscosity, and by oxidation or contamination with solid or tarry mate-

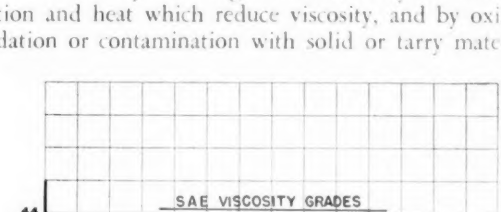
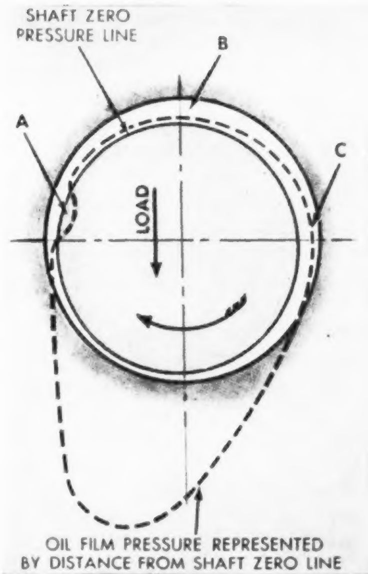


Figure 19 — Oil pressure-viscosity relationship for a gasoline engine. See page 28 for discussion.



Courtesy of Eaton Manufacturing Company

Figure 20 — Section of a journal bearing with polar diagram of oil film pressure. **See page 33 for discussion.**

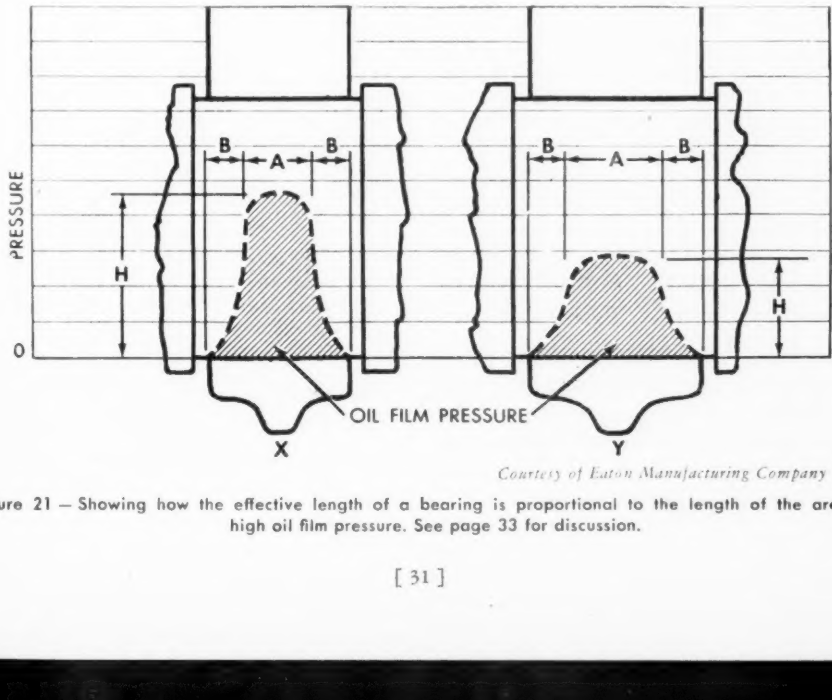
rials which tend to increase viscosity. However, if dilution, oxidation or contaminant products present in engine oil change the viscosity of the oil the equivalent of more than one SAE grade, then either oil change periods are too extended or improper operation or performance of the engine are indicated. From Figure 19 it may be noted that a change of one SAE grade in the engine with a normal pump resulted in less than a one psi change in oil pressure which cannot be detected readily on most engine oil pressure indicators.

Increasing viscosity grades in a rotating journal and bearing combination results in an increase of heat generation from oil film shearing with increasing oil viscosity. Accordingly, this increases the temperature of oil returned to the bulk oil to be supplied to the bearing. Therefore, the net effect is a tendency for the changes in oil temperature to offset the viscosity effect shown in Figure 19 which was determined at a constant temperature. Of course the inclusion of an oil cooler in the system would alter this condition.

Effect of VI On Engine Oil Pressure

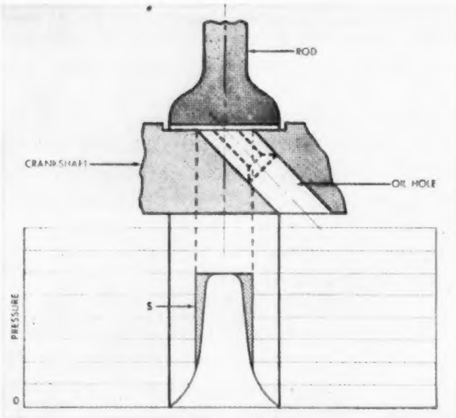
A question is usually raised regarding the effect of VI on oil pressure since a low VI oil decreases in viscosity more rapidly than a higher VI oil as temperature increases. In the discussion above it has been pointed out that viscosity is the controlling factor, therefore, it would seem that differences in VI would have an effect. However, in the temperature ranges associated with normal engine operation the viscosity difference between oils of different VI's in the same grade are relatively small, therefore, there should be very little difference in engine oil pressure. This is substantiated by the following data obtained in an automotive gasoline engine at constant 180°F. jacket and 190°F. oil temperatures with oils of approximately equivalent 210°F. viscosity but different viscosity index.

Oil	*P-3C*	*N-4A*	*P-3*	*N-5*
VI	102	63	97	0
Engine Speed, RPM	Oil Pressures, PSI			
1200	32.3	33.3	32.0	32.2
1800	35.4	35.8	34.0	36.0
2400	36.4	37.0	36.5	37.2
3000	38.0	38.8	38.0	39.0



Courtesy of Eaton Manufacturing Company

Figure 21 — Showing how the effective length of a bearing is proportional to the length of the area of high oil film pressure. **See page 33 for discussion.**



Courtesy of Eaton Manufacturing Company

Figure 22 — Showing modern practice of drilling an angular hole from the main bearing to feed the rod bearing. See page 33 for discussion.

It is apparent that VI had little or no effect on the engine oil pressure under normal operating conditions. Similar data, shown below, from a railway Diesel engine again show that the lubrication system did not recognize VI during normal operation at 240°F. oil temperature and 200°F. jacket out temperature.

Oil	*Vis. SUS at 210°F.*	*VI*	*Oil Pressure, PSI*
A	68.9	0	47
B	77.0	56	54
C	75.2	61	50
D	79.5	86	51
E	80.0	92	47

It is recognized that at low oil temperatures, such as when starting a cold engine, differences in VI are reflected by differences in oil pressure since engine oil pressure is a function of viscosity. Under these conditions of engine operation the importance of using the proper viscosity oil to relieve peak cold oil pressures and increase flow is apparent.

On starting a cold engine (oil cold) a marked increase in oil pressure may be noted. This is a viscosity effect. As the engine warms up the oil viscosity is reduced and oil pressure should drop to normal. If this does not occur in a reasonable length of time, immediate action should be instituted to determine the cause of the high pressure. Conversely, if soon after starting an engine, oil pressure is not indicated the engine should be shut down and the cause of no oil pressure determined.

Detergent or Dispersive Oils

It has been noticed that a change in oil pressure sometimes results after changing from a straight mineral oil to a detergent or dispersive type. It is well known that these oils have a reasonable purging action in an engine. Loosened deposits may clog oil pump screens and intake lines causing reduced oil flow and reduced pressure or they may get by the pump and clog oil galleries increasing oil pressure by limiting oil flow. By cleaning deposits from worn bearing and pump clearances, the increased leakage can cause an oil pressure decrease. It is obvious that some precaution should be exercised in changing from a straight mineral to a detergent and/or dispersive type oil in a used engine. None of these effects were ever noted by changing to a detergent or dispersive oil in newly overhauled or new engines that were in good mechanical condition.

As mentioned previously the oil in a pressure lubricated engine also serves as a coolant. It is therefore desirable to have as high a rate of flow as possible consistent with the amount of oil control provided by the power section of the engine, otherwise excessive consumption can result. An increase in flow resulting from increased bearing clearances can result in lowered oil pressure and may appear to be beneficial from its heat carrying standpoint, however, this may be greatly offset by loss of oil control and the attendant high oil consumption.

Oil Pressure Indications of Engine Malfunctioning

The above discussion and points set forth are

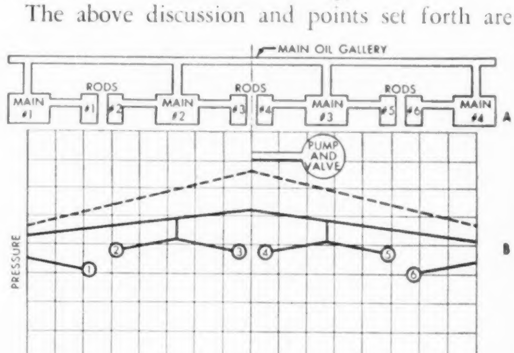


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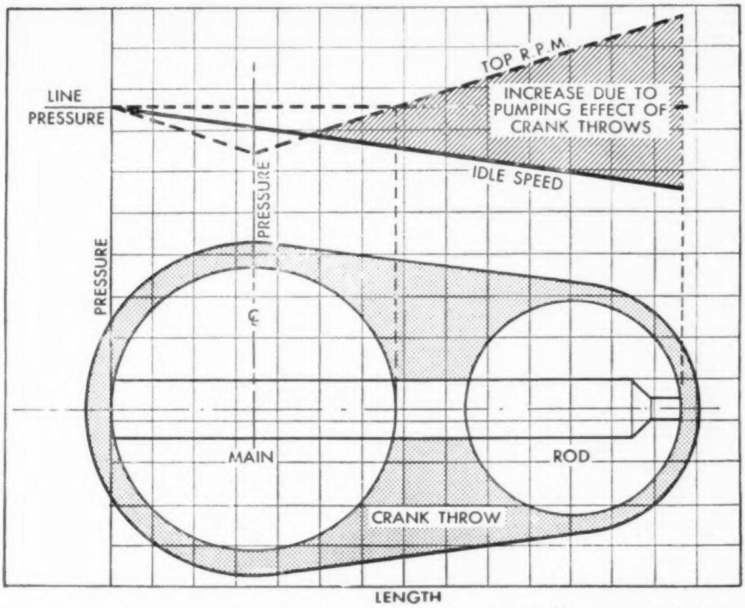
Figure 23 — Schematic diagram of the primary lubrication system of an engine. See page 34 for discussion.



Courtesy of Eaton Manufacturing Company

Figure 24 – Showing pressure relationship between main and rod bearings. See page 34 for discussion.

not intended to mean that changes in oil pressure are detrimental to engine operation. Most engines in good mechanical condition, operating under a fixed set of conditions and using a specified viscosity grade oil will show a definite oil pressure indication. If, after continued use under the same conditions a change in oil pressure is noted, it is a good indication that something in the lubrication system is not normal. The most common change is that of reduced oil pressure. The amount of reduction that can be tolerated is dependent on the engine design. In some engines relatively small reductions may lead to malfunctioning while in others rather large reductions may not necessarily cause failure although this is not considered to be a desirable condition.

Knowing the engine characteristics with regard to engine oil pressure it is possible to associate engine mechanical malfunctions with deviations from normal engine oil pressure. Some of these malfunctions and their effect on oil pressure are shown listed on page 36.

Trends In Lubrication System Design*

Bearings

One of the most important individual problems in a modern automotive engine lubrication system is the journal bearing. A section of a journal bearing with a polar diagram of the oil film pressure is shown as Figure 20.

With gravity feed the oil is usually introduced between A and B and must pass through the region of larger journal clearance to arrive at C where it is available to form the oil wedge. Much oil may be lost as leakage while passing through the large clearances.

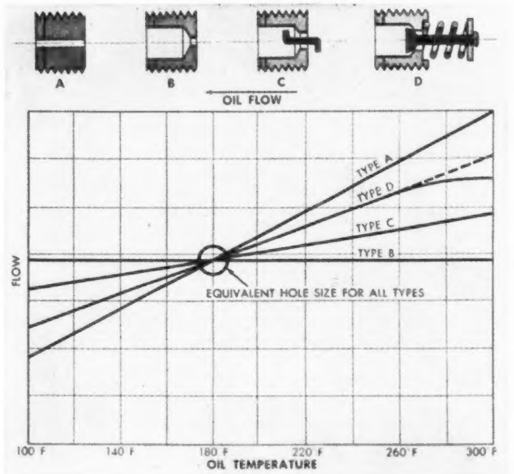
This oil may not only be lost for lubrication but may be ineffective for cooling. In a pressure system oil may be introduced nearer point C, thereby reducing end leakage and having a maximum amount of oil contacting the bearing surfaces for cooling.

The effective length of a bearing is proportional to the length of the area of high oil film pressure (length A Figure 21). The ineffective length caused by end leakage is constant regardless of bearing length, therefore, any change in bearing length affects the high load area directly. Although the change is a small per cent of total bearing length, it can be an appreciable per cent of the high load area and greatly affect the load carrying capacity. This is illustrated in Figure 21 wherein two bearings of different lengths carrying the same load are shown. The oil gradient B is equal, however, compare H and note the difference in bearing pressure.

From the standpoint of bearing cooling it is essential that end leakage oil does some cooling by passing over the journal. However, in some designs the use of an annular groove combined with a longitudinal dirt slot may carry the oil away before it can exert any beneficial cooling.

The modern practice of drilling an angular hole from the main bearing to feed the rod bearing, as

*October, 1946, "Wil-Rich Forum", Published by Eaton Manufacturing Company.



Courtesy of Eaton Manufacturing Company

Figure 25 — Secondary lubrication can be provided by bleed holes. See text below for discussion.

shown in Figure 22, may, in the case of a narrow bearing, remove most of the end seal. The dotted lines are a suggested practice to increase the end seal, adding greatly to the cooling oil contact as well as directing the oil pressure to the loaded bearing oil wedge rather than to its gradient leakage edge. The shaded area S in Figure 22 represents an oil loss resulting from the large oil feed hole.

Some recent studies have shown that the load carrying capacity of a bearing is greatly increased when the load rotation is opposite to the shaft rotation as is the case during the highly loaded portion of the cycle of a rod bearing. Also a bearing can support an instantaneous load much greater than the average load under which it will fail.

Lubrication System Pressures

Figure 23 shows (A) a schematic diagram of the primary lubrication system in an engine; (B) the effect on pressure of locating the pump and bypass at the center of the oil gallery; and (C) the effect on pressure with the pump at one end and relief valve at the opposite end of the gallery. The dotted lines indicate pressure gradients for cold oil temperature while solid lines are for normal temperature conditions. In the former (B) peak cold oil pressures are controlled by a reduction in flow while in the latter (C) consideration must be given to line size to keep cold oil pressure and resulting pump drive torque within a reasonable limit. However, there is a gain in oil flow because of the increased pressure.

The pressure relationship between main and rod bearings is shown as Figure 24. The centrifugal pumping effects of the crank throws increase the oil pressure at the rod bearing, thereby increasing the flow of oil. The oil pressure at the rod bearing increases as the square of the speed to approximately 30 psi at 4000 RPM. Bearing temperature varies almost directly with speed. Therefore, an increase in oil flow occurs as the cooling requirements increase.

Secondary Lubrication

Secondary lubrication is usually provided via bleed holes; four general types are shown with corresponding flow curves as Figure 25. The orifice size should be determined for the most normal operation temperatures, however, type A orifices are usually determined at cold oil temperatures with a resultant increase of flow at the higher temperatures. The knife edge orifice type B has a discharge nearly independent of viscosity. By using smaller hole diameters more oil is available in the primary system under cold temperature conditions.

A design to decrease danger of dirt fouling, by means of the action of a free piece of wire, is shown as type C. In this type, hole size can be reduced in view of its annular nature and less danger of clogging whereas in the straight hole type, size of hole is limited to drill size rather than determined by actual oil flow. A shut-off type used to confine oil to the primary system up to a certain pressure, which is usually low, and may be designed to give variable flow, is shown as type D. This type is often used to isolate the oil filter from the engine at low speeds or it may be incorporated in the engine relief valve as shown in Figure 26.

It cannot be emphasized too strongly that a squirt hole of 3/32", for ease of production, can actually rob the vital engine bearings of a major portion of engine oil pump flow. Even a 1/16" hole is usually too large and a loss through it represents three times as much oil flow as is given to any vital engine bearing. A larger pump capacity cannot always rectify this squirt hole loss. At engine idle with

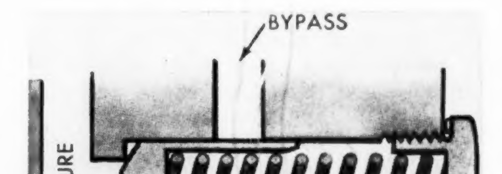


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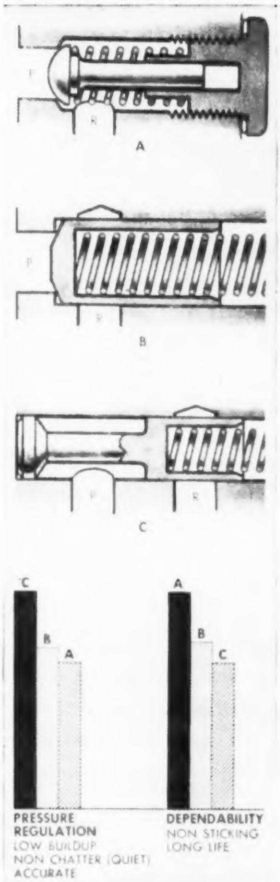
Figure 26 — Showing a type of shut-off valve. See text above for discussion.



Courtesy of Eaton Manufacturing Company

Figure 27 — Three types of relief valves. See text below for discussion.

hot oil a 6 GPM at 2000 RPM pump will deliver 0.3 GPM. A 10 GPM at 2000 RPM pump will deliver 0.5 GPM at idle. The 1/16" hole will flow 0.3 GPM at 50 psi, therefore, pressure cannot be achieved with a 6 GPM pump. With a 10 GPM pump, even if the engine should be tight enough to maintain pressure with a 0.2 GPM flow to the bearings, less than half flow is available for the bearings. There seems to be no compromise from a small properly proportioned oil hole in combination with an efficient pump.

Relief Valves

There are two general types of relief valves, those which by-pass to the sump and those which return oil to the pump intake. These are also divided into wet or dry types, depending on whether they drain dry during periods of non-operation.

With a relief valve incorporated in the pump housing it is usually most convenient to by-pass to the pump inlet. Therefore, the valve should be of the wet type and non-sticking to prevent air leakage and loss of pump prime.

To assure priming, by-passing to the sump would be preferable. The valve can then be wet or dry and sticking not be a priming hazard. If possible the return should be below the oil level to decrease aeration and foaming.

Three types of valves that have proven satisfactory are shown as Figure 27. Selection, variation and combination of these designs should be to meet the desired characteristics. Types A and B are low pressure types having a larger seating area exposed to pressure when open than when closed, thereby causing the valve to hunt under critical conditions; this may be disastrous in a high pressure system. Type C overcomes this, however, in view of the close fits required, the oil must be kept exceptionally clean.

Aside from dirt, lacquering is a common cause of relief valve sticking. Type C is the worst offender under these conditions. By increasing OD clearances type B can be improved while type A is loose and is least affected. Type C is best for accurate pressure control and quietness.

Oil Pumps

Engine designers have increased lubrication points as they have become lubrication conscious but with modern auto styling oil pans have been reduced in size, thereby limiting the size of oil pump that can be installed. Therefore, the problem cannot be solved by merely increasing pump size. In the past large manufacturing tolerances were employed resulting in large pump clearances despite the fact that in a capillary-sealed pump the leakage varies as the cube of the clearance. The pump was selected so that the discharge equaled requirements.

It is usually desired to reach relief pressure by 1000 engine RPM (500 pump RPM) and if possible maintain full pressure at idle or at a time when pump efficiency is lowest. From Figure 28 it can be seen that the leakage of a conventional gear pump is constant at a given pressure so that at a lower RPM leakage forms a larger portion of the total discharge. Thus a pump with 80% efficiency at 1000 RPM (pump) has 20% efficiency at 250 RPM. The dotted lines in Figure 28 show the performance of a smaller pump with good low speed efficiency. It was designed to give the same flow as the first pump at the RPM at which the relief valve opens. As a result it gives more oil up to the point at which the valve opens and less by-passed oil beyond this point. This action can also be applied to giving more oil throughout the low speed range without increasing top flow and pump drive power.

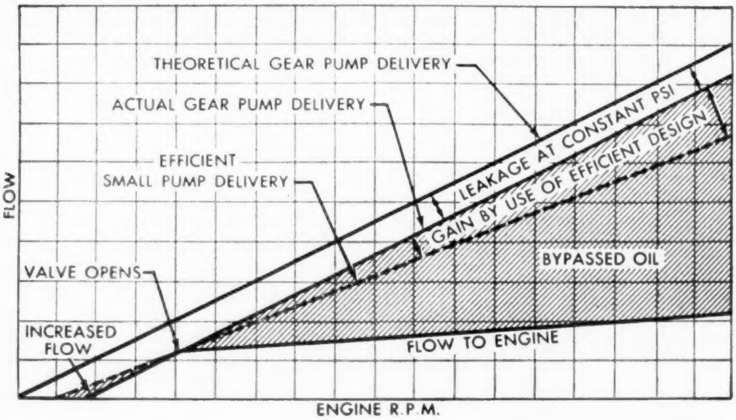


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Figure 28 — Oil pump performance data. See page 35 for discussion.

CONCLUSION

The lubrication system is the life line of an engine; any malfunctioning in this system may lead to serious engine failure. Therefore, it is a primary requisite in engine operation to maintain the lubrication system in good condition and to use the proper lubricating oil to obtain maximum service from the engine. Engine oil pressure is used to indicate oil flow in the lubrication system, but at times the significance of variations in engine oil pressure is not completely clear, therefore, before conclusions are drawn regarding changes in engine oil pressures, the characteristics of the engine with respect to oil pressure and/or previous experience on the engine in this respect should be ascertained.

CONDITIONS AFFECTING OIL PRESSURE

Some factors affecting oil pressure and their indication on the oil pressure gauge are as follows:

	Pressure Gauge Indications
1. Faulty gauge	High or low
2. Clogged line to gauge	No movement or delayed action
3. Clogged oil pump screen	Low
4. Faulty oil pump	Low or erratic
5. Excessive main, con rod, camshaft or rocker arm bearing clearances	Low
6. Clogged full flow filter (provided by-pass is inoperative)	Low
7. Insufficient oil filter restriction	Low
8. Ineffective oil cooler, depending on type, may keep oil too cold or provide insufficient cooling	High or low
9. Crankcase oil level just at or below oil pump pick-up	Erratic, then low, eventually none
10. Use of excessively low viscosity oil	Low
11. Use of excessively high viscosity oil	High
12. Excessive fuel dilution	Low
13. Excessive contamination	High
14. Clogged oil passages on the pressure side	High
15. Enlarged squirt holes	Low
16. Improper setting or failure of pressure relief valve	High, low or erratic
17. Oil lines or line fittings broken, cracked or loose	Low
18. Oil pump drive gears stripped or drive broken	Low, erratic or none
19. Oil too cold	High
20. Restrictions in oil pan or oil too viscous to keep oil pump intake supplied	Low or erratic
21. Oil pump pick-up stuck high	Low or none

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